CONFIDENTIAL

Energy-Efficient Fume Hoods

(Low-Flow Fume Hoods)

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1. ABSTRACT

Laboratory buildings have very high energy intensities. Conditioning of the make-up air to be exhausted by fume hoods uses most of the energy beyond what is required for technical apparatus and lighting. Based on 1 million installations in the US, we estimate that fume hoods in California alone provide a peak-power challenge of 1 GW to utilities. For the U.S., the energy consumption just for fume hoods is large, we estimate about 0.27 EJ (0.27 Quadrillion Btu) per year.

Providing a safe environment for all personnel is the primary objective in the design of HVAC systems for laboratories. Fume hoods require high exhaust air flows, which often lead to higher outside air flows than required for the occupants alone. In other words, the energy needed to condition and to transport air is often determined by fume hoods. To reduce a laboratory's energy needs, the exhaust air flow for fume hoods needs to be reduced or fume hoods have to be supplied with unconditioned auxiliary air.

Auxiliary air causes uncomfortable conditions, and reducing exhaust air leads to lower face velocity, which, with conventional fume hoods, increases the risk of pollutants spilling from the fume hood into the laboratory.

While the performance of a conventional fume hood depends on an even distribution of air velocity in the face of the hood, the energy-efficient low-flow fume hood design works on the principle of an air supply with low turbulence intensity in the face of the hood. The air flow supplied displaces the volume currently present in the hood's face without significant mixing between the two volumes and with minimum injection of air from either side of the flow.

If the face of the hood is protected by an air flow with low-turbulent intensity, the need to exhaust large amounts of air from the hood is largely reduced. Based on preliminary experiments, we estimate that exhaust air flow reductions of 75 to 80% are possible without a decrease in the hood's containment performance.

This report reviews the literature dealing with design criteria for conventional fume hoods and describes a new fume hood design that reduces conditioned make-up air while maintaining the level of protection offered by a conventional hood.

2. INTRODUCTION

Fume hoods significantly affect the amount of energy consumed in laboratory buildings. Make-up air quantity, exhaust capacity, ductwork size, fan power needs, boiler size, and chiller capacity are often determined by the number and type of fume hoods installed. Fume hoods represent an enormous life-cycle cost because they are operated 24 hours a day, every day of the year. Every 300 cfm (510 m^3/h) of exhaust air requires approximately one ton (3.52 kW_{th}) of refrigeration (Cooper 1994). A single 6-ft-wide fume hood with a sash opening height of 2.5 ft (opening 1.52 m by 0.76 m) can create cooling loads of 4.2 tons (14.77 kW_{th}).

Figure 1 shows the fume hood nomenclature used in this report.

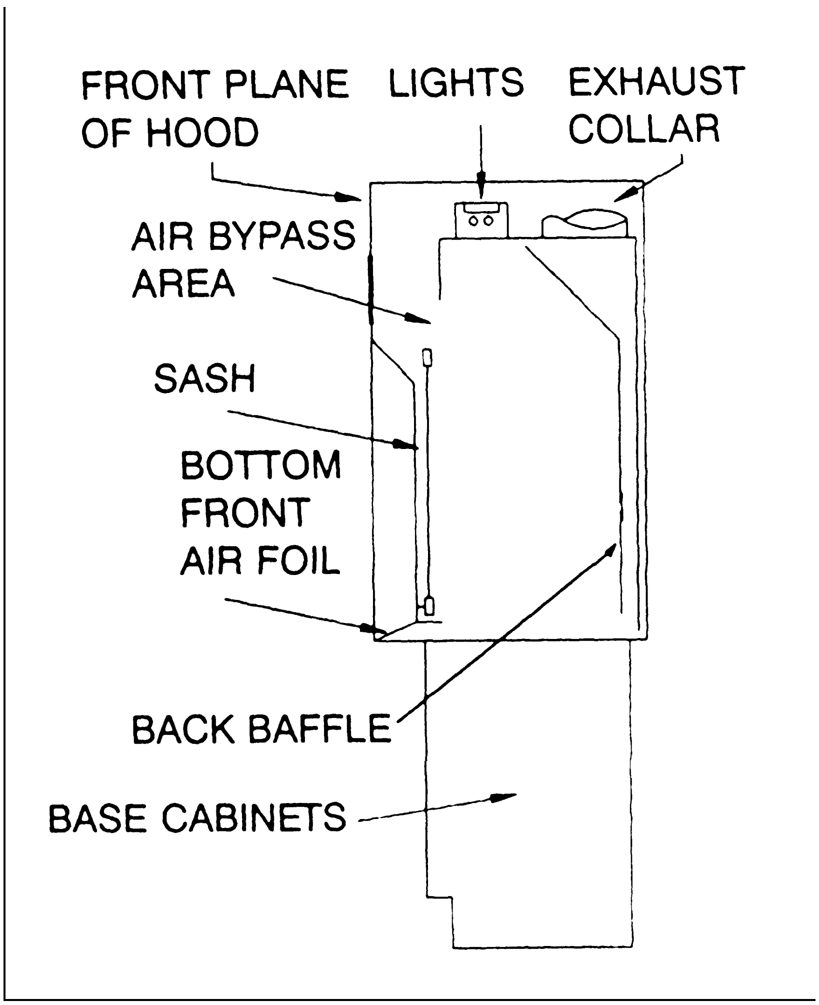

Figure 1: *Fume Hood Nomenclature (Saunders 1993)*

A fume hood is a “ventilated enclosed work space intended to capture, contain, and exhaust fumes, vapors, and particulate matter generated inside the enclosure. It consists basically of side, back, and top enclosure panels, a floor counter top, a sash, and an exhaust plenum equipped with a baffle system for air flow distribution” (ASHRAE 1995). The purpose of chemical fume hoods is to draw fumes within the work chamber away from the worker, so that inhalation of contaminants is minimized. The concentration of contaminants in the breathing zone must be kept as low as possible and should never exceed the threshold limit value (TLV) (Cooper 1994).

The first fume hood, used by an alchemist, was a fireplace. Fireplaces and fume hoods share a number of features. Like fireplaces, early fume hoods had fairly tall chimneys, with thermal updrafts resulting from thermal buoyancy caused by fire. During the Industrial Revolution, the gas-burning rings used to increase draft were replaced by mechanical fans. The first major improvement after fume hoods were provided with sashes was the addition of the back baffle system, which forces air to be exhausted from the hood's working surface area as well as from the top canopy area (Saunders 1993).

In the 1940s, the Atomic Energy Commission had the Harvard School of Public Health develop equipment for fume hood operation and safety. High Efficiency Particulate Arrestors (HEPA) filters and fume hood entrances designed to take air flow patterns into account were the result. Saunders (1993) says that, despite the claims of hood manufacturers, little has been added to basic hood design since then.

Although hood face velocity of 50 feet per minute (fpm) (0.25 m/s) was originally considered adequate, the value increased over time to 150 fpm (0.75 m/s) to “improve” hood safety. Only when a research project sponsored by the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) produced a procedure for establishing fume hood performance were face velocities reduced to the range of 60 to 100 fpm (0.3 to 0.5 m/s) (Caplan and Knudson 1978). This research formed the basis of ASHRAE Standard 110-1985, which is the quantitative method for evaluating of the performance of laboratory fume hoods.

3. DESIGN CRITERIA FOR CONVENTIONAL FUME HOODS

3.1 GENERAL

Containment of contaminants in a conventional fume hood is based on the principle of a directed (inward) air flow in the face of the hood. The open face corresponds to the area below the sash at the front of the hood through which air enters (ASHRAE 1985).

For safe fume hood operation, effective air circulation in the laboratory is essential. Depending on the ceiling height, Bell et al. (1996) recommend six air changes per hour (ach) of outside air for a safe B-2 occupancy laboratory. For laboratories that routinely use hazardous material, such as carcinogens, 10 to 12 outside air changes per hour are recommended. The "rule of thumb" [1cfm per ft^2 (17 m^3/h per m^2)] (Bell 1997) provides 6 ach for laboratories with a ceiling height of 10 ft (3.05m).

A fume hood with a face opening of 5 ft by 2.5 ft (1.52 by 0.76 m) and a face velocity of 100 fpm (0.5 m/s) exhausts 1,250 cfm (2,080 m^3/h), which would provide sufficient exhaust for a laboratory space of 1,250 ft^2 (116 m^2).

A fundamental goal of energy engineers is to reduce the amount of exhaust air to the lowest safe level because conditioning of make-up air is very energy intensive. Bell and collaborators (1996) state that surprisingly few codes stipulate the actual amount of exhaust for laboratory-type facilities.

3.2 FACE VELOCITY

Recommendations for face velocity range from 75 fpm (0.37 m/s) for materials of low toxicity (Class C: TLV > 500 ppm) to 130 fpm (0.65 m/s) for extremely toxic or hazardous materials (Class A: TLV < 10 ppm) (Cooper 1994). In general, industrial hygienists require minimum face velocities of 100 fpm (0.5 m/s) for hoods with open sashes.

However, as shown above, face velocity recommendations have changed over time. In the 1970s, recommendations for face velocity moved from 50 fpm (0.25 m/s) to 150 fpm (0.75 m/s) and higher. Face velocities higher than 125 fpm (0.63 m/s) can create significant turbulence inside the hood, causing fumes to spill into the laboratory (Monsen 1989). The literature reveals there is no direct relationship between face velocity and containment level; many factors are responsible for the effectiveness of a fume hood.

3.3 OTHER INFLUENCES ON CONTAINMENT

In addition to the hood design, the position of the worker with respect to the air flow direction has a significant influence on the air flow patterns in the hood, and particularly in the face of the hood. Air flows surrounding the body standing in front of a hood create a region of low pressure downstream of the person. This region, which is deficient in momentum, is called the wake. It disturbs the directed air flow in the face of the hood and can cause the contaminant to spill (ASHRAE ACGIH,1995).

In general, a hood's overall “box leakage factor” (sash leakage and box leakage) correlates strongly with turbulence intensity. The National Institute of Health (1996) found that sash leakage is dependent on laboratory air flow patterns. The turbulent fluctuation in air velocity generated in the room is carried into the hood by the general flow of air.

Therefore, a hood's performance is affected by the hood's location with respect to doors, supply air outlets and areas with foot traffic. Saunders (1993) shows that even the highest proposed hood face velocity is smaller than the air velocities created by door openings [175 to 450 fpm (0.83 to 2.25 m/s)] or people passing the hood [260 to 450 fpm (1.30 to 2.25 m/s)]. Supply air diffuser can create air velocities in the vicinity of the hood that are higher than the design face velocity.

A hood's position in relation to other hoods influences the hoods' performance. The National Institute of Health's study (1996) suggests placing fume hoods on the same wall at least 4 ft (1.22 m) apart, preferably in corners. Hoods on opposite walls perform well, but best performance is achieved when fume hoods are installed on perpendicular walls. In any case, maximizing the distance between two hoods on the one hand and the supply air grille on the other hand provides the best performance. For more details about laboratory design, see Bell et al. (1996).

3.4 Construction Details of Conventional Fume Hoods

The size of a fume hood describes its outside dimensions. The width of the interior work chamber is found by subtracting the size of the two side walls from the total width. Therefore, a 6 ft (1.83 m) fume hood with side walls of about 6 inches each (0.15 m) has an interior work chamber width of 5 ft (1.52 m). The side walls have considerable width because they provide an aerodynamically shaped entrance to the hood chamber and contain mechanical and electrical services.

Hood depth includes the thickness of the outside shell and can vary from 32 to 37 inches (0.81 to 0.94 m). The depth of the work space depends on the design of the hood's air foil and its the back baffle (Saunders 1993). This leaves a work area that is approximately 21 inches (0.53 m) deep. The dimensions of the work space within the fume hood should be determined by the worker's needs. Using a hood that is larger than needed wastes initial costs, energy and operating costs (Cooper 1994).

The most important aerodynamic design feature of the fume hood entrance is the air foil, which prevents the formation of turbulent air flow in the hood's working area. The equivalent measure for the hood exhaust is the back baffle. Optimum hood air flow design “sweeps” the work area, but also prevents contaminated air from a vortex that forms above the open sash from re-entering the air flow coming through the hood's face (see **Figure 2**).

If room air flow patterns create a cross draft to the hood, air flow in the face might change direction. If contaminated air within the hood is drawn into the flow reversal, the hood has become unsafe.

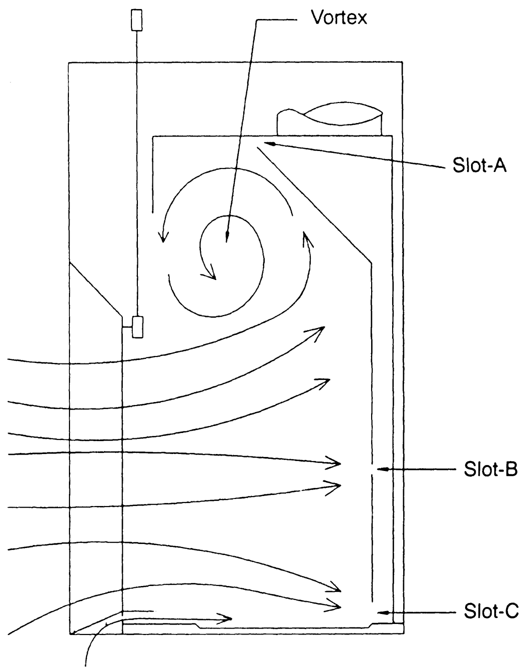

Figure 2: *Air Flow Pattern inside a Fume Hood (Saunders 1993)*

Movable sashes offer additional safety that is not available from an open-faced hood. Sashes come in either vertical or horizontal arrangements. A vertical sash can provide an open face area of 100%, but there are some limits on the open area with a horizontal sash. Therefore, the air flow requirement for fume hoods equipped with a (non-removable) horizontal sash is smaller than for their counterparts with a vertical sash arrangement.

4. ENERGY SAVINGS STRATEGIES

4.1 GENERAL

As discussed earlier, fume hood exhaust contributes significantly to the energy consumption of laboratory-type facilities. The high energy consumption caused by exhaust air flows is a result of the need to condition the make-up air and to move it through the building's air handling system. In the past, we have seen three different attempts to reduce energy costs from fume hoods:

1. applying a hood diversity factor in calculating of the building's make-up air volume
2. providing unconditioned air to fume hoods (Auxiliary Air Fume Hood)
3. controlling the amount of air flow as a function of the hood's sash location (VAV-Fume Hood)

"Hood diversity" is the assumption that not all hoods are being used simultaneously. By assuming a "hood diversity factor," the engineer assumes that a certain percentage of hoods are shut off (Moyer and Dungan 1987, Varley 1993). Because the supply air handling system has to supply (transport and condition) only the amount of air actually being exhausted, switching off hoods saves energy. For safety reasons, we do not suggest switching off hoods. HVAC engineers often use a 75% diversity factor to design the supply air system; however, observations in laboratories have shown hood diversity factors as low as 40% (Saunders 1993).

4.2 EXISTING ENERGY-EFFICIENT FUME HOOD DESIGNS

4.2.1 Auxiliary Air Fume Hood

Auxiliary air fume hoods supply unconditioned (or "less-conditioned") air near the top and front of the hood sash. Therefore, the amount of conditioned room air exhausted by the hood is reduced. However, the less-conditioned air (up to 95% of the exhaust) often causes thermal discomfort in winter when outside air is cold [preheating is provided only up to 55 F (13^{o}C)] or in summer when outdoor humidity levels are high. Auxiliary air can also adversely impact experiments. Anecdotal evidence suggests that auxiliary air fume hoods are not highly regarded by the laboratory personnel (Saunders 1993).

In addition to the problems related to the thermal condition of auxiliary air, the system itself presents some engineering challenges. First, the auxiliary air must be of reasonably uniform velocity across the discharge area, and second, the discharge velocity must not exceed the face velocity of the fume hood by more than 20% (Saunders 1993). Balancing the two separate supply air systems for the laboratory can be another engineering challenge.

Auxiliary air fume hoods reduce the amount of energy used to condition make-up air. They reduce operating costs by saving energy and reduce first costs by permitting installation of downsized heating and cooling equipment. However, they do not reduce fan energy consumption because they do not change the amount of exhaust air.

Auxiliary fume hoods are part of the energy-efficiency and safety features built into the new laboratory building of the University of Oklahoma (Sarkey Energy Center 1997). Although the University of Oklahoma praises auxiliary fume hoods, others report the risks involved in using them. Coggan (1997) mentions the high probability "that the air flow will vary with variation in duct static and hood flow, which adds an element of unpredictability to the system." He reports that hoods with downward velocity components of more than 10 fpm (0.2 m/s) will have a "guillotine effect" on the air stream entering the hood, and remarks that "it is worth noting that Public Works Canada prohibits the use of hoods with auxiliary air." Therefore, it does not come as a surprise that Coggan suggests "not (to) use hoods with an auxiliary air supply. (If) it exists already, convert it to a safe design with the proper face velocity control."

4.2.2 VAV Fume Hoods

Conventional constant-volume fume hoods are not at all "constant face-velocity hoods." The exhaust air fan removes approximately the same amount of air no matter what the sash position. If the sash is lowered, the face velocity increases and might reach unsafe levels (see **3.2 Face Velocity**). In order to provide sufficient air flow to dilute contaminants in the hood and to avoid air whistling when the sash is closed, a bypass is provided above the sash.

Constant face-velocity fume hoods are equipped with variable air volume exhaust fan and automatic controls. Fume hoods equipped with VAV regulate the amount of exhaust from the hood to obtain a relatively constant face velocity. The exhaust air flow can be controlled by sensing:

a) the face velocity,
b) the sash position, or
c) the pressure between the inside of the hood and the room.

VAV systems not only control the exhaust but also the amount of make-up air, by means of multiple dampers (Maust and Rundquist 1987). VAV fume hoods are safer than conventional hoods, because the face velocity stays constant independent of the sash position, at least in theory. Therefore, installation of VAV hoods is often a safety issue rather than an energy-saving measure.

User discipline (or automatic controls to determine whether a person is present at the hood (Bentsen 1997) is necessary for the VAV system to save energy. For the lowest exhaust air flow, the design criterion should be the necessary dilution level, e.g., the lower explosive limit (Saunders 1993). Although many papers address air flow control issues (Wenz 1989, Lacey 1989, Maust and Rundquist 1987, Rabiah et al. 1989), only one publication provides measured energy savings data. According to Bell et al. (1996), 60 to 70% energy reductions can be achieved from the reduced need of make-up air (reduced air conditioning and fan power) with a VAV system.

We have not found any information about peak-power savings related to VAV fume hoods, but we assume it is of equal magnitude to savings during normal daytime use.

5. FUME HOOD TESTING

5.1 INTRODUCTION

Fume hoods are tested according to ANSI/ASHRAE 110-1995 "Method of Testing Performance of Laboratory Fume Hoods." This standard tests but does not specify performance. "The desired hood performance should be defined as a result of a cooperative effort of such people as the user, the chemical hygiene officer, and the applications engineer. It should be noted that the performance test does not give a direct correlation between testing with a tracer gas and operator exposures.... The performance test does, however, give a relative and quantitative determination of the efficiency of the hood containment under a set of specific, although arbitrary, conditions" (ANSI/ASHRAE 1995).

The method consists of three tests:
1. flow visualization
2. face velocity measurements
3. tracer gas containment

"The flow visualization and the face velocity tests should always precede tracer gas testing for a thorough evaluation of hood performance. This portion of the standard could be used in the testing and balancing of new facilities and periodic tests of many hoods at a large facility. The full procedure (visualization, face velocity, and tracer gas) is a quantitative measurement of a hood's containment ability and is useful for hood development and rigorous evaluation of hood performance" (ANSI/ASHRAE 1995).

Depending on the conditions of the test, the rating might be "as manufactured" (AM), "as installed" (AI), or "as used" (AU). Manufacturers' catalogue data typically reflect

5.2 FLOW VISUALIZATION AND FACE VELOCITY

Flow visualization shows a hood's ability to contain vapors. "The test consists of both a small local challenge and a gross challenge to the hood. The intent of this test is to render an observation of the hood performance as it is typically used" (ANSI/ASHRAE 1995). The hood's air foil and overall containment are tested as smoke is released on the side walls and on the floor parallel to the hood face and six inches (150 mm) behind the face of the hood. In addition, smoke is released at the back of the hood to detect air flow reversal or lack of air movement. Additional smoke releases show whether all smoke is carried to the back of the hood and exhausted. "If there is visible smoke flow out of the front of the hood, the hood fails the test and will receive no rating" (ANSI/ASHRAE 1995).

For the gross visualization challenge, a large volume of smoke is released in the center of the sash opening on the work surface 6 inches (150 mm) inside the rear edge of the sash. A steady visible release of smoke from the hood indicates failure.

The face velocity measurement procedure requires that the velocity be measured in a grid pattern. Velocity readings are taken with a calibrated anemometer fixed at the center of the grid spaces. Readings should be integrated over a period of at least five seconds. The average velocity is calculated and the highest and lowest readings are noted. No criteria are specified for appropriate velocity or its distribution.

5.3 TRACER GAS TEST PROCEDURE

The most comprehensive performance test for a fume hood is a tracer gas test. This test requires injecting a tracer gas (SF6) with a given release rate (4 L/min) into the hood. The tracer gas injector is described in the ANSI/ASHRAE standard. The injector is placed at different positions (left, center, right), each 6 inches (150 mm) from the hood face. A tracer gas sensing probe is positioned in the breathing zone of a mannequin placed in front of the hood. Detector readings are observed and recorded at least every 10 seconds for 5 minutes. The performance rating of the hood is then recorded either AUyyy, AIyyy, or AMyyy, where yyy equals the average of the tracer gas concentration in ppm during the five-minute test. A test rating of AU 0.5 indicates that the hood controls leakage into the laboratory to 0.5 ppm at the mannequin's breathing zone sensing point. The test does not indicate whether a rating represents good or a bad performance.

5.4 EVALUATION OF THE STANDARD

The fume hood test procedures described by the ANSI/ASHRAE standard cover visual tests, face velocity tests, and tracer gas tests. Only the tracer gas test provides results that show a hood's containment ability, but this test is usually only performed once. Paradoxically, the face velocity test, which does not directly correlate to the hood's ability to contain contamination, is performed more often, because it is the least expensive of the test options allowed by the standard. However, regular use of tracer gas testing would be more effective to insure safe hood operation.

6. LOW-FLOW FUME HOOD

6.1 LOW-FLOW FUME HOOD PRINCIPLE

The low-flow fume hood is a new design that combines innovations with features of earlier fume hood designs, particularly those intended to reduce turbulence. Because the person standing in front of a hood is a source of turbulence for the air flow through the hood's face, several attempts have been made to compensate for the impact of a flow obstacle in front of the hood.

The air vest was invented for use with large paint spray hoods (Gadgil et al. 1992). The vest supplies air in front of the operator of the spray hood, which creates a positive pressure field. This high pressure prevents development of a wake and therefore ensures clean air to the operator's breathing zone.

A system with an effect similar to that of the air vest is the auxiliary air fume hood. Here, the air to be exhausted from the fume hood is supplied above the operator's breathing zone on the outside of the hood. Therefore, a person standing in front of the hood has only a minimal impact on the flow.

The low-flow fume hood design also uses an air supply that is placed between the person in front of the hood and the hood face. The performance of a conventional fume hood depends on an even distribution of air velocity in the face of the hood, but the low-flow fume hood design works on the principle of an air supply with low turbulence intensity in the opening of the hood. The air flow supplied displaces the volume currently present in the hood's face without significant mixing between the two volumes and with minimum injection of air from either side of the flow. This principle will provide a protective layer of clean air between the contaminated low-flow fume hood and the laboratory room. Because this protective layer of air will be free of contaminants, even temporary mixing between the air in the face of the fume hood and room air, which could result from short-term pressure fluctuations in the laboratory, will keep contaminants contained in the hood.

6.2 Proof of Concept

6.2.1 First Attempt

The following describes a mock-up of a conventional fume hood which was used to check whether the concept would work. We constructed a frame (made of PVC pipe with a rectangular area) which encloses the face of a conventional fume hood (see **Figure 3**).

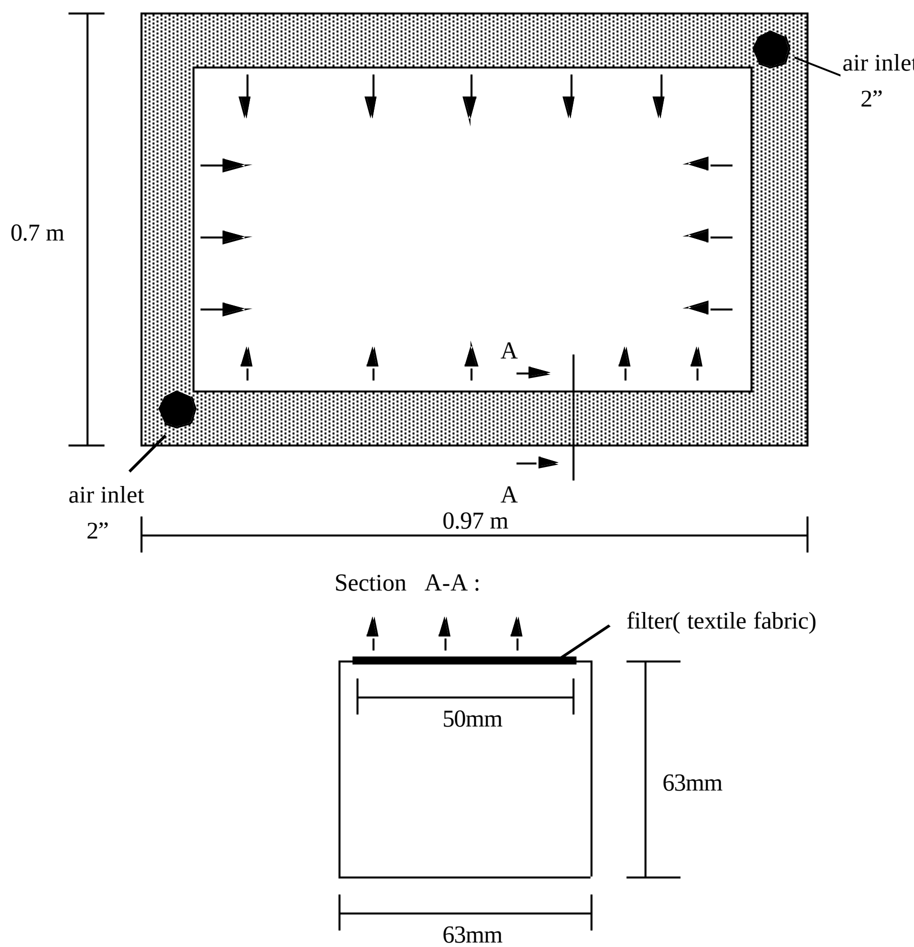

Figure 3: *Details of the frame*

Figure 4: *Fume hood operating with 33% exhaust air flow; but without additional supply air from frame. Fog spills out of the hood*

Figure 5: *Fume hood operating with 33% exhaust air flow; but with additional air supply from frame. Spills are being prevented*

The frame was cut open towards the center of the face. These open areas are covered with fabric, that allows the air to flow at low velocity and low turbulent intensity towards the center of the face of the fume hood. At the air outlet, air flow is perpendicular to the flow found in conventional hoods. The supply air taken from the laboratory itself (NOT auxiliary air flow!) builds a protective buffer zone between the volume of the hood and the laboratory space.

The exhaust air flow in the mock-up can be modified by a damper placed in the exhaust duct above the hood. The fan on the roof of the building only exhausts air from this hood. Before inserting the frame, the open face of the hood with the sash fully elevated was 0.97 m wide and 0.70 m high.

As the frame was not fully integrated into the hood design, air was supplied to the frame by flexible duct at two points only (lower left corner and upper right corner of the frame). This arrangement caused high turbulences within the pipes forming the frame, and consequently some uneven air velocities for the four supply air surfaces were observed.

The design exhaust air flow for the conventional hood (with a face opening reduced by the frame) at 100 fpm (0.5 m/s) is 994 m^3/h. For our tests we reduced the exhaust air flow to approximately 33% of the design air flow. For flow visualization we used an ultrasonic humidifier, which produced fog and ejected it with a small velocity into the hood. The fog supply was directed towards the open face. **Figure 4** shows the flow visualization result for the reduced exhaust without air supply by the frame. Due to the higher density of the cool fog, spills can be observed at the bottom of the hood.

If part of the make-up air is supplied by the frame (approximately 50% of the exhaust air), no spills of fog are visible (see **Figure 5**). **Figure 6** shows a close-up of Figure 7. The fog crawls from the center of the working space of the hood towards the lower air outlet of the frame and is in the face of the hood effectively being displaced by the supply air.

This experiment shows that a fume hood can contain contaminants even with low exhaust air flows if sufficient countermeasures are taken. Because of the limited supply air of the frame, the low-turbulent air supplied by the frame mainly protects the critical locations of the fume hood — the edges of the face. We expect that higher supply air flows will provide further potential to reduce exhaust air flows.

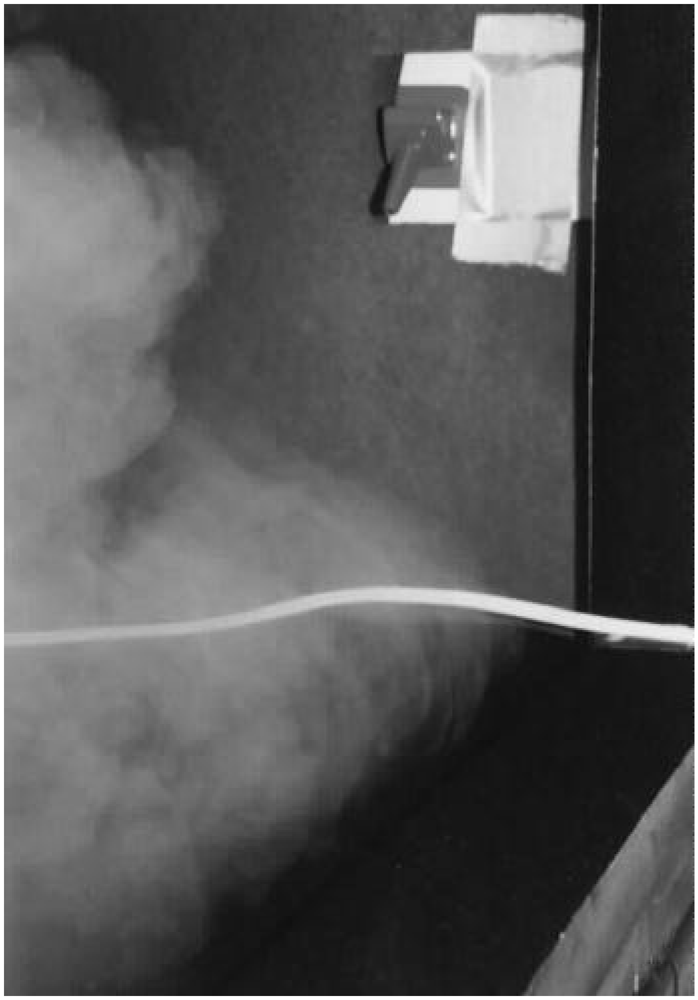

Figure 6: *Fume hood operating with 33% exhaust air flow; but with additional supply air from frame. Detail of* ***Figure 5***

6.2.2 Second Attempt

A fume hood mock-up built of foam board was used to test the containment capabilities of this technology. The mock-up represents a 5 ft (1.524 m) hood with side walls each being 6 inches (0.152 m) wide. The face is 4 ft (1.219 m) wide and 2.5 ft (0.762 m) high. The depths of the hood is 33 inches (0.838 m). Part of the air to be exhausted by the hood is being supplied in the face of the hood. Air grilles covered by wire mesh are located inside and outside the sash both on the upper and lower part of the face. The air grilles are fed by an upper and lower air plenum, which are supplied with room air by axial fans.

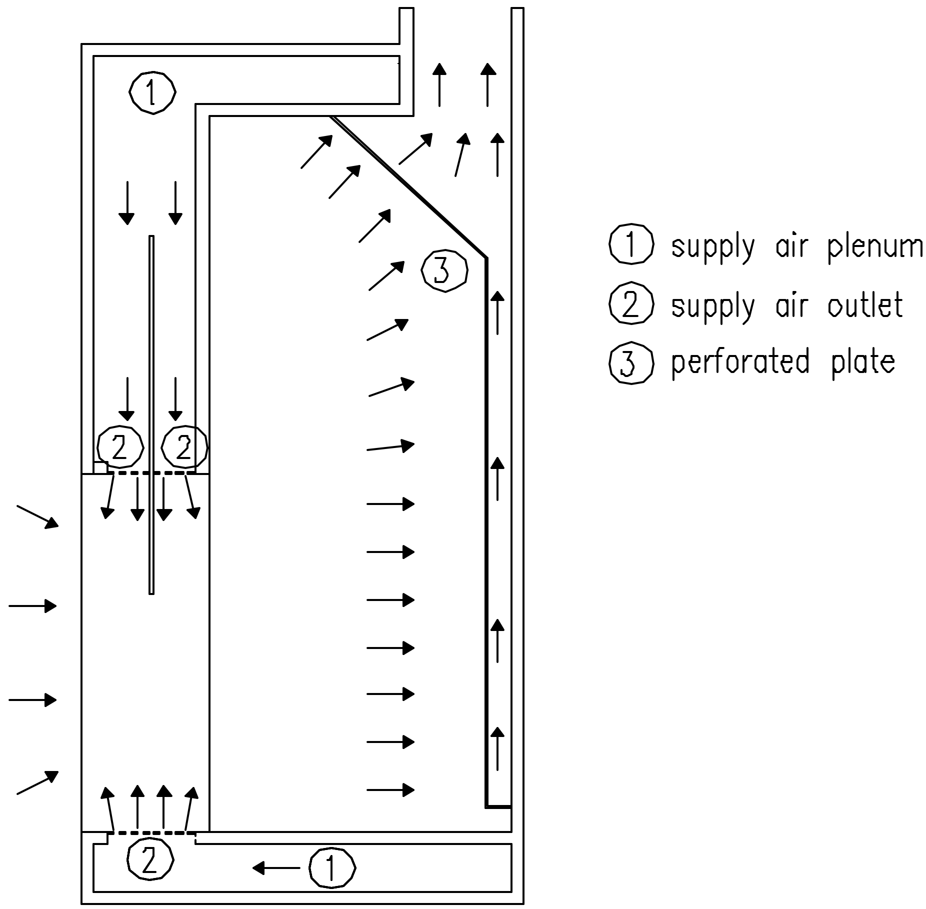

Figure 7: *Design principle of energy-efficient low-flow fume hood*

To be able to provide the air evenly over the width of the hood's face with low turbulence intensity at a very low pressure drop, the air flow patterns within the hood's supply path was optimized. Positioning of fans, flow straighteners, air flow guides in the plena, etc. are based on experimental studies performed on a model of the hood's supply air system. The conventional three-slot back baffle design was replaced by a back baffle with holes distributed in a particular pattern.

Figure 7 shows the design principles of the mock-up.

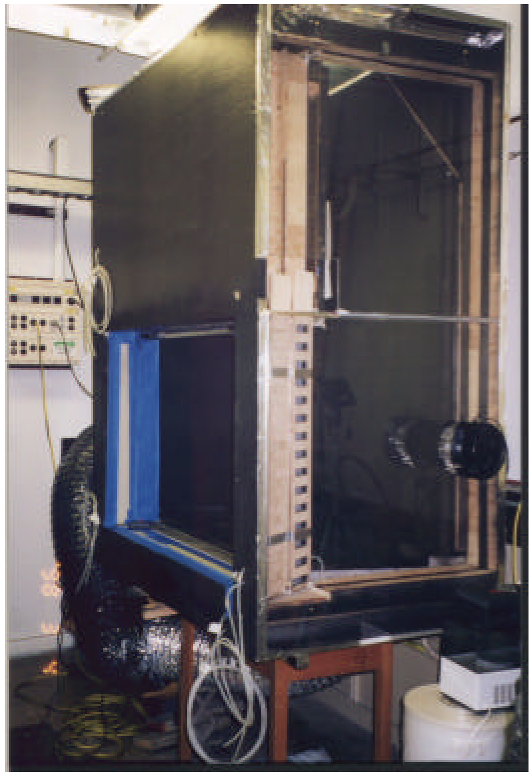

Figure 8: *Mock-up of energy-efficient low-flow fume hood with outlets for forced air supply on all four sides of the face. Vertical outlets and lower horizontal outlet outside the sash are taped-off.*

6.2.3 Testing the Mock-up

To evaluate the performance of containment of the mock-up, several test were performed. Initial tests were based on fog (denser than air) and theatrical smoke (lighter than air). Both visual tests showed good containment of the hood. To visualize the air flow patterns in the face of the hood, smoke was injected into the supply air plenum serving the lower horizontal outlet. **Figure 9** shows how the supply air emerges perpendicular to the outlet; further away from the outlet the air flows towards the back of the hood. As we can assume that the air flow emerging from the upper outlets shows a similar flow pattern, the open face is guarded by the two air curtains provided by the forced supply outlets. Only about 10% of the exhaust is passing from the room through the face into the hood.

Figure 9 also shows how little mixing takes place between the air supplied to the hood and the room air. We assume that a similar mixing behavior could be observed on the inside of the hood.

Figure 9: *Smoke injected in the lower supply air plenum shows the air flow patterns of the lower part of the face.*

The ultimate test for a fume hood however, is the tracer gas test described by ANSI/ASHRAE 110-1995. The tracer gas containment tests were performed by Ratcliff & Associates, an independent tester and member of the ASHRAE 110 standards committee.

The tracer gas test requires to inject tracer gas (pure SF_6) with a given release rate (4 L/min) into the hood. The tracer gas injector is described in the standard. The injector is placed at different test positions (left, center, right), each 6 inches (0.150 m) from the hood's face. The tracer gas sensing probe is positioned in the breathing zone of a manikin placed in front of the hood. The detector readings shall are observed and recorded for 5 minutes with a reading taken at least every 10 seconds.

The performance rating of the hood is then recorded either 4 AUyyy, 4 AIyyy, or 4 AMyyy, where yyy equals the average of the tracer gas concentration in ppm during the five-minute test and the number "4" indicates the volumetric rate of tracer gas injection. Depending on the conditions of the test, the rating might be "as manufactured" (AM), "as installed" (AI), or "as used" (AU). Manufacturer's catalogue data typically reflects "AM" test ratings. A test rating of 4 AU 0.5 indicates that the hood controls leakage into the laboratory to 0.5 ppm at the manikin's breathing zone sensing point when 4 L/min pure SF_6 are injected. ANSI/AHI Standard Z9.5 (1992) section 5.7 gives an indication whether a test result is acceptable or not. Because of the location of the mock-up in the laboratory space the tester decided that our hood arrangement reflects the category "as installed." Therefore, the five minute average of the SF_6 concentration in the breathing zone should not exceed 0.1 ppm.

Figure 10: *Preparing the tracer gas test*

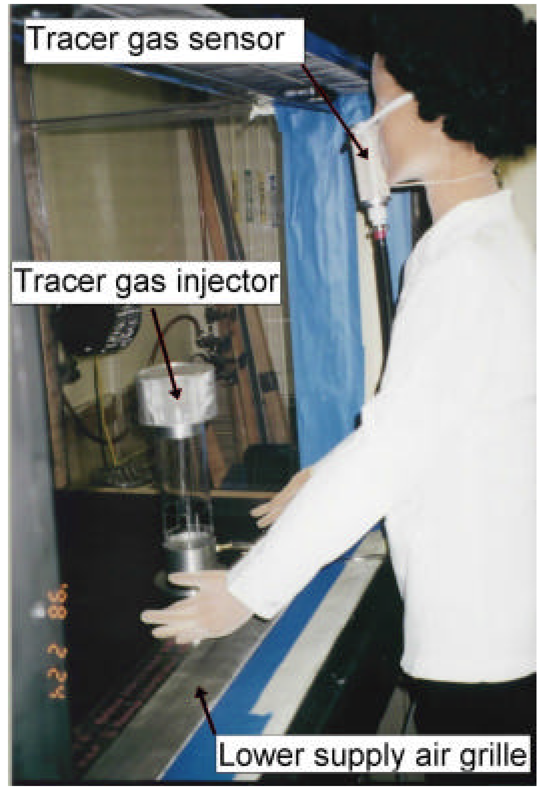

Figure 11: *Test setup for tracer gas test. Tracer gas injector and dummy are in the "right" position (see Appendix).*

The hood was operated at exhaust air volumes of about 25% (of a conventional hood with a face velocity of 100 fpm (0.5 m/s)) and a forced supply air volume of 90% of the air exhaust. Under these conditions, the mock-up passed the tracer gas tests for all three positions, center, left, and right. A patent application was filed in April 1998.

With the air volumes further reduced the mock-up failed. For detailed test results, please see Appendix.

6.3 Savings Potential

If it were possible to protect the face of the hood by means of the supply of a low-turbulent intensity air flow, the need to exhaust large amounts of air from the hood would be largely reduced. Based on preliminary experiments, we estimate that exhaust air flow reductions up to 80% are possible without a decrease of containment performance.

Monsen (1989) states that there are more than one million existing fume hoods which have to be retrofitted to comply with regulations. We use this number to calculate the energy implication low-flow fume hoods could have for California. We assume that the hood population per capita is 50% higher in California than for the whole U.S. and that all hoods are 6 feet (1.80 m) wide, have an open sash of 30 inches and operate at 100 fpm (0.5 m/s). Since energy is being used to transport (fan power) and to condition the air, we also assume that the average laboratory air system has a pressure drop of 1200 Pa.

With all these assumptions we calculate the number of hoods in California to be retrofitted to 173,157 with 1,250 cfm (2125 m^3/h) each. Thus, the exhaust air flow from those fume hoods would be 216 million cfm (367 million m^3/h). For a fan/motor arrangement with an efficiency of 0.6 this translates into a fan power requirement of 204 MW or 1.79 TWh consumption per year.

For the thermal load we use the rule of thumb that 300 cfm require 1 ton of refrigeration. Therefore, a 6-foot fume hood requires 4.2 tons of refrigeration (14.8 kW_{th}). With a chiller COP of 3.5 this translates into 4.2 kW electrical load. For all 173,157 Californian hoods this calculates to 725 MW electrical load. Adding the fan power for air transport and the power need for air-conditioning the make-up air we calculate a peak-power requirement of 929 MW. If we can provide the same containment with less than 33% of the exhaust air, more than 600 MW could be saved.

The objective of the future work is to support, accelerate, and augment private and public sector efforts to improve the energy efficiency of the U.S. commercial building stock. The potential savings just for laboratory fume hoods are large, $7.7\ 10^{12}$ Wh electrical energy for reduced fan operation alone ($83\ 10^{15}$ J primary energy). Reduced energy for conditioning exhaust air amounts to approximately $114\ 10^{15}$ J primary energy. Together that amounts to almost 0.2 EJ (0.2 Quads) of primary energy per year. These savings are not likely to be achieved without a major effort to address the technical, institutional, financial, and educational barriers to improving the energy efficiency of laboratory fume hoods. The purpose of this project is to optimize the design of an energy-efficient low-flow fume hood, to demonstrate the savings potential, and to develop the necessary technology transfer mechanisms to overcome these barriers.

6.4 CFD Investigation of an Energy Efficient Fume hood

6.4.1 Introduction

In this chapter, results from CFD simulations are used to examine the air flow in an experimental fume hood under four operational scenarios. At this stage, only the lengthwise plane of symmetry of the fume hood is represented in the simulations. However, since the hood construction does not vary greatly in the depth direction, i.e., slot outlets and exhaust, this representation will provide very useful information.

The overall geometry and problem description are illustrated in **Figure 12**, and the four operational scenarios are illustrated in **Figure 13**. A small section of the room has been included in the simulations, as shown in **Figure 12**, so that the flow into the hood working area can be calculated, as opposed to supplying this flow as a boundary condition.

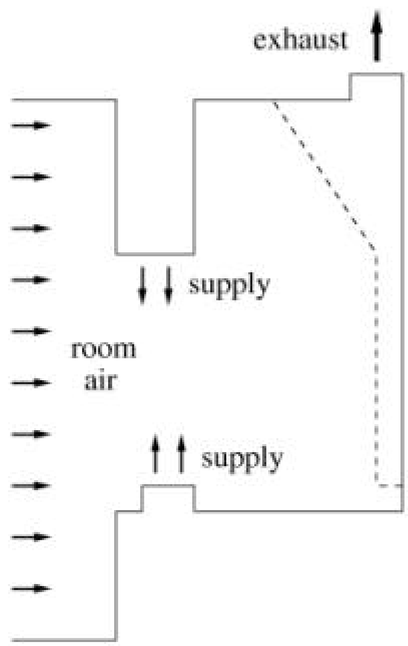

Figure 12: *General problem setup for the fume hood simulations*.

In the first three scenarios, the fume hood is in normal operation with a total air flow of 262 cfm. Each supply outlet provides 45 % of the total flow. The remaining 10 % of the air are drawn from the room. The directional distribution of the supply air coming from the top and bottom outlets is altered in each case to examine the effect on the air flow structure in the hood. In the fourth scenario, the supply air from the outlets is turned off. All of the air flowing through the hood comes from the room, instead of just 10 % as in the other cases. In this case, the air flow through the hood is adjusted to provide an average face-velocity of 100 ft/min in the working area (the standard for conventional fume hood operation), which gives a total air flow of 1046 cfm. The last case provides a basis for comparison of the fume hood under conventional operation.

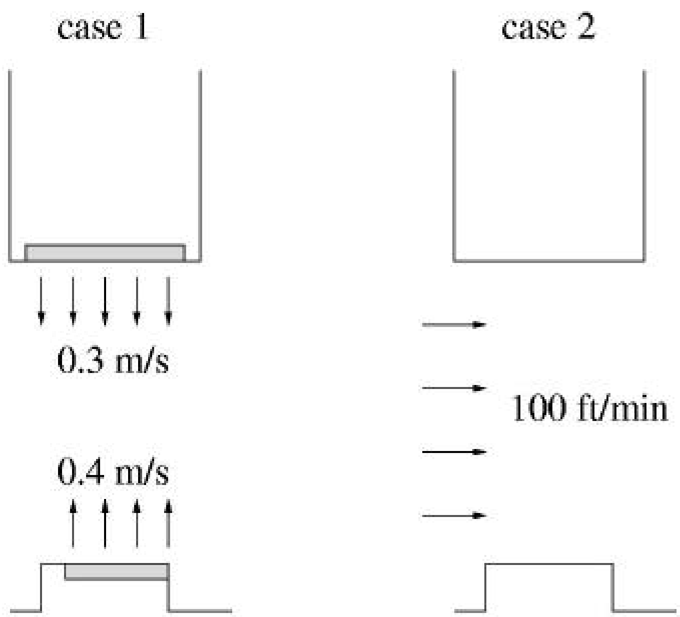

Figure 13: *Air flow boundary conditions at hood outlets for different scenarios*.

6.4.2 Mathematical Formulation

Simulations are performed with a commercial CFD package, which utilizes a control volume formulation. The air flow is calculated via the time-dependent, two-dimensional Navier-Stokes equations, given in Equations 1-3. The flow is isothermal and has constant material properties. Due to the complex nature of the flow, a direct steady-state solution technique could not provide a converged solution. Therefore, a transient technique was used, in which time-marching was continued until the steady-state solution was achieved. All information provided in this report is from steady-state solutions.

$$\frac{\partial r}{\partial t} + \frac{\partial r u}{\partial x} + \frac{\partial r v}{\partial y} = 0 \qquad (1)$$

$$\frac{\partial ru}{\partial t} + \frac{\partial ruu}{\partial x} + \frac{\partial ruv}{\partial y} = -\frac{\partial p}{\partial x} + m\frac{\partial}{\partial x}\left[2\frac{\partial u}{\partial x} - \frac{2}{3}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y}\right)\right] + m\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) \quad (2)$$

$$\frac{\partial rv}{\partial t} + \frac{\partial ruv}{\partial x} + \frac{\partial rvv}{\partial y} = -\frac{\partial p}{\partial y} + rg + m\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) + m\frac{\partial}{\partial y}\left[2\frac{\partial v}{\partial y} - \frac{2}{3}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y}\right)\right] \quad (3)$$

Prescribed velocity profiles are used as boundary conditions at the supply outlets and room far-field. In case 1, each supply outlet provides 45 % of the total air flow. In case 4, the supply outlets are turned off and 100 % of the air is drawn from the room. At the exhaust, an outflow boundary condition is used, which sets variable gradients equal to zero. At all solid surfaces the air velocity is set to zero.

A challenging aspect of modeling the fume hood was how to represent the perforated baffling, a unique feature of the hood design, which must be represented properly in order to achieve accurate simulation results. It was decided to represent the baffling as a porous media with a pressure drop equal to an inertial loss term. This is a common technique for representation of perforated plates in CFD simulations. The governing equations for air flow in the porous region are given by Equations 4 and 5. The inertial loss factor, *C*, was determined through laboratory experiments.

$$\frac{\partial p}{\partial x} = C_x \left(\frac{1}{2} r u |u| \right) \quad (4)$$

$$\frac{\partial p}{\partial y} = C_y \left(\frac{1}{2} r v |v| \right) \quad (5)$$

In these experiments, a small section of perforated board, like that used for the baffling, was connected to a flow device and subjected to various flow rates which cover the range experienced during normal operation of the hood. Measurements of the pressure drop across the board were taken at about six different air flow rates, allowing the loss factor to be calculated as a function of flow. This process was carried out for three configurations of the perforated board: one with all of the holes open, a second with half of the holes open, and a third with one quarter of the holes open. In the fume hood baffling, the number of perforations per unit area varies by design, so this information is needed for an accurate representation.

6.4.3 Results

Simulations are performed for the fume hood under the four operational scenarios discussed previously. In the first three cases, an average pressure drop of about 18 Pascals across the baffling was predicted. Laboratory measurements taken with a handheld pressure gauge show about the same pressure drop during normal operation. This verifies that an important feature of the real fume hood has been accurately captured in the simulations. Simulation results, hereafter, will be presented graphically through plots of air flow velocity vectors and streamlines.

Figure 14 shows two plots of the air flow in the fume hood under operating scenario 1. The plot on the left shows contours of the stream function, with blue as the clockwise vortex and red as the counter-clockwise vortex. Contours of a given color reveal streamlines, tangents to the flow velocity vectors, shown in the plot on the right. Large flow rates occur in regions where the distance between streamlines is small (rapid transition between colors). Streamlines that form closed loops indicate zones of recirculating air. Two large recirculation zones appear inside the hood, one near the floor of the working area (blue) and another higher up toward the exhaust (red). These recirculation zones are also evident in the vector plot. It can be seen in the vector plot that the supply air leaves the outlets vertically, but is quickly turned toward the rear side of the hood. The supply air, then, flows between the two recirculation zones and spreads along the baffling before it is drawn to the exhaust side and removed.

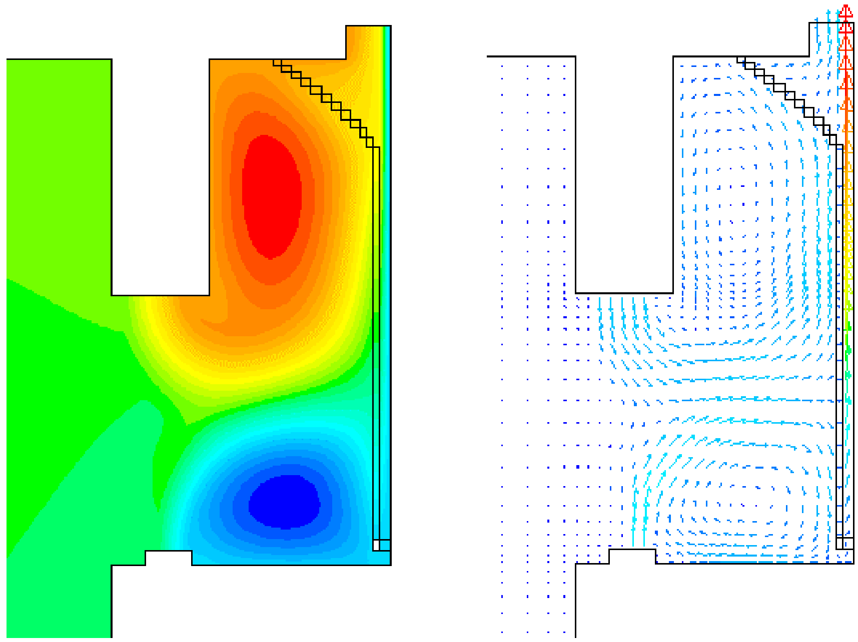

Figure 14: *Filled contours of the stream function, streamlines, (left) and air flow velocity vectors (right) for case 1*.

During ideal operation, there would be a displacement flow through the hood, which would uniformly sweep contaminants from the working area to the exhaust. In this case, however, there are areas with high and low flow velocities and large recirculation zones. The recirculation zones prevent ideal operation and should be eliminated to provide a more uniform flow.

Figure 15 shows plots of the air flow for the final scenario, case 4. In this case, supply air from the outlets has been turned off and the total flow has been increased to 1046 cfm, four times that of the previous cases. The two recirculation zones are still apparent in the contour plot, but are slightly smaller. The streamlines and velocity vectors show that the flow in most of the working area is fairly uniform, as desired.

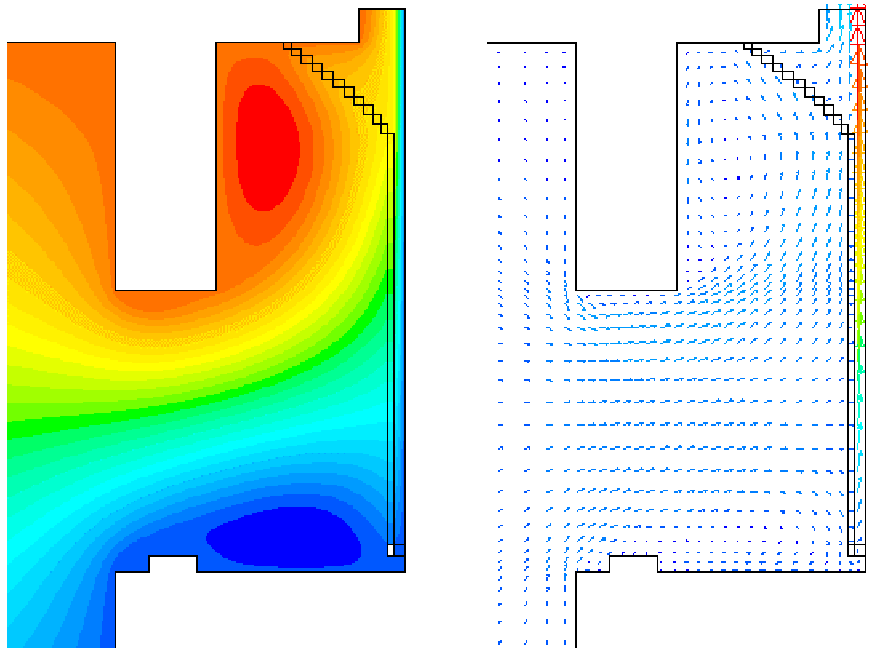

Figure 15: *Filled contours of the stream function, streamlines, (left) and air flow velocity vectors (right) for case 4.*

The room air flows uniformly through the working area to the baffling and is drawn toward the exhaust. This configuration appears to provide a displacement flow, the ideal flow condition. Of course, this comes at a huge operating cost. The goal is to make the low-flow fume hood operate like a hood with a conventional flow rate.

6.4.4 Conclusions

Results from this study provide detailed information about the air flow in the experimental fume hood. Changes made in the supply flow distribution clearly affect the structure of the air flow in the hood, suggesting that this technique could be used to successfully design the desired flow through the hood. Further experimentation will be required to achieve the optimal design. In future work, it would be desirable to include an air contaminant in the simulations to examine the flow's transport characteristics. This would provide the most important information about the effectiveness of the fume hood.

7. FUTURE WORK

Improving the energy efficiency of laboratory fume hoods through new design for new and existing units offers a large potential for energy savings in the next 5 to 15 years in the United States. Replacement of the one million hoods by more energy-efficient ones is slow. Consequently, most of the existing stock will remain in use for the next 30 to 40 years. Therefore, we do not focus only on the optimization of new hoods, but also work on retrofit options.

7.1 OPTIMIZE DESIGN

Two-dimensional computational fluid dynamics (CFD) models will be used to optimize the design of the shape and location of supply air outlets within the face of the hood. CFD will also be used to design the supply air plenum to minimize turbulence intensity and pressure drop. While low turbulence intensity in the supply air plenum is important to "displace" air in the face with supply air without significant mixing with air in the hood, a low pressure drop reduces the amount of fan energy needed to supply the air. Two-dimensional computational fluid dynamics (CFD) models will also be used to design the back-baffle to optimize the containment capability of the hood and to minimize the pressure drop for the exhaust flow.

7.2 BUILD PROTOTYPE

Build and test prototype with optimized supply air outlets and back-baffle design. The prototype will be designed taking CFD modeling results into consideration. Containment tests will be performed. The prototype design will be optimized based on results from flow visualization and tracer gas measurements. Final adjustments will be necessary because the optimization modeling is based on two-dimensional while airflow in the hood is of three-dimensional nature.

7.3 TECHNOLOGY TRANSFER AND DISSEMINATION

Information dissemination will be highlighted by presentations at technical conferences, publications in professional and popular journals as well as interaction with industry representatives. LBNL will reach out to fume hood manufacturers to market the energy-efficient low-flow fume hood.

7.4 DESIGN RETROFIT OPTION

Based on airflow studies performed for new fume hood design, retrofit options will be designed. CFD modeling will be performed to study the airflow patterns inside the supply air duct and the supply air plenum and to minimize both, turbulence intensity and pressure drop of supply air outlet.

7.5 BUILD RETROFIT OPTION

A kit to convert existing conventional fume hoods to energy-efficient low-flow fume hoods will be built based on CFD modeling results. The conversion kit will be installed in existing conventional fume hoods in the field. Containment tests will be performed.

8. ACKNOWLEDGEMENTS

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APPENDIX A: PEAK-POWER SAVINGS POTENTIAL

APPENDIX B: ENERGY SAVINGS POTENTIAL

APPENDIX C: TEST RESULTS

10. APPENDIX A: PEAK-POWER SAVINGS POTENTIAL

Calculation of Peak-Power Savings Potential:

Air Flow per Hood:

$$\dot{V} = L \times H \times v = 5\,ft \times 2.5\,ft \times 100\,fpm = 1250cfm$$

Number of Hoods to be retrofitted:

$$n = Hoods_{total} \times \frac{Population_{California}}{Population_{USA}} \times Hood\_Density = 1{,}000{,}000 \times \frac{29{,}760{,}000}{257{,}800{,}000} \times 1.5$$

$$n = 173{,}157$$

Air flow of Hoods to be retrofitted

$$\dot{V}_{total} = n \times \dot{V} = 173{,}157 \times 1250cfm = 216{,}250{,}000cfm$$

Fan Power Consumption of these Hoods

$$P = \frac{\dot{V} \times \Delta P}{h} = \frac{367{,}200{,}000m^3/h \times 1{,}200Pa}{0.6} = \frac{102{,}000m^3/s \times 1{,}200N/m^2}{0.6}$$

$$P = 204{,}000{,}000Nm/s = 204MW$$

Fan Energy Consumption for these Hoods

$$Q = P \times 8760 = 204{,}000{,}000W \times 8760h = 1{,}787{,}040{,}000{,}000Wh = 1.79TWh$$

Thermal Peak Load for Chiller

$$P_{thermal} = exhaust\_air \times load\_factor = 1{,}250cfm/hood \times \frac{1ton}{300cfm} = 4.167ton/hood$$

Electrical Peak-Power for Chiller

$$P_{el,chiller} = \frac{P_{thermal}}{COP} = \frac{4.167ton/hood \times 3.52kW/ton}{3.5} = 4.19kW/hood$$

$$P_{el,chiller,total} = P_{el} \times n = 4.19kW/hood \times 173{,}157hoods = 725{,}527kW$$

Electrical Peak-Power for Chillers and Fans

$$P_{el,total} = P_{el,chiller,total} + P_{el,fan,total} = 725MW + 204MW = 929MW$$

11. APPENDIX B: ENERGY SAVINGS POTENTIAL

a) Costs
downsizing of the compressor, the boiler, the air handler, the duct work
operational costs for heating, cooling, fan power

b) energy
a conventional hood of 5x2.5 face has:
1) exhaust volume of 1250 cfm
2) cooling peak of 1 ton per 300 cfm => 4.2 tons per hood
3) fan power of 1 W/cfm (Title 24?) => 1.25 kW per hood (supply and exhaust)

assumptions:
4) fans run 8760 hours a year => 1.25 kW x 1,000,000 hoods x 8760 hours
=> 1.1 10^13 Wh = 3.9 10^16 J
5) heating and cooling for the whole
US over the year is 50% of p
=> 4.2 tons/hood x 1,000,000 hoods x 0.5 x 8760 hours x 3.5 kW/ton
=> 6.4 10^13 Wh = 2.3 10^17
6) makes together 0.27 EJ; 75% savings = 0.2 EJ

Regarding hood size: standard conventional hoods have an opening of 4 ft wide by 2.5 ft high, however, a large number (we don't know specifics) are much wider. 6ft and 8 ft wide benches are very common. Therefore, we work here with an average of 5 ft.

12. APPENDIX C: EVALUATION OF FUME HOOD CONTAINMENT USING ASHRAE 110 TRACER GAS TESTS

for

LAWRENCE BERKELEY LABORATORIES

sponsored by:

LAWRENCE BERKELEY LABORATORIES
One Cyclotron Road
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prepared by:

RATCLIFF & ASSOCIATES
Laboratory Fume Hood Testing Services
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Ratcliff & Associates Project No. 00160
LBL Subcontract No. 6471337

March 1998

ASHRAE 110 TRACER GAS TEST REPORT

Lawrence Berkeley Laboratories
February 23-24, 1998
Ratcliff & Associates Project 00160

Subcontract No. (LBL) 6471337
Contact Person: Dr. Helmut Feustel
Description of Fume Hood

experimental proprietary design; low-flow fume hood with supply air from top and bottom edges of face perimeter. Hood is of simple construction, not highly aerodynamic, and intended only as a proof of concept.
Sash full open at 29”. Face width = 48”.

Description of Test Procedure

Basic tracer gas test without sash movement effects.
No face velocity tests performed due to low face velocities of unique design.
Dry ice procedure of ASHRAE 110 Appendix used and videotaped by LBL.

Acceptability Level

0.1 ppm or less for 5 minute average at all 3 mannequin positions, based on ANSI/AIHA Standard Z9.5 (1992), Section 5.7. The As-Installed or As-Used designation is appropriate for this case since the room conditions were not carefully controlled as would occur at a hood manufacturer laboratory.

Deviations (if any) from ASHRAE 110 procedure

Horizontal distance from sash to center of probe = 4.5 inches rather than 3 inches due to hood design of upper face area. Mannequin forehead was against hood and could not be moved forward more.

Exception Report

1. inspection showed a small pressure regulator leak at SF6 supply. Supply was moved to a negatively-pressured chamber in room which exhausted at far end of fume hood room.
2. larger leak found in fume hood exhaust ductwork, repaired satisfactorily. After these 2 leaks addressed, background generally returned to near zero concentration following each 5 minute SF6 test. The tracer gas detector was re-zeroed slightly as necessary before each 5 minute test.

Results Description.

Table 1 summarizes test plan and results, indicating the mannequin positions, run number, average and maximum tracer concentrations, and a Pass/Fail designation. The runs are grouped to show the effects of various parameters.

The fume hood passed the ASHRAE 110 test with the initial setup configuration: exhaust flow setting of 72 Pa and supply flow settings of 2.2 Pa and 2.3 Pa for the upper and lower supply vents. Exhaust and supply flows set by LBL. The three mannequin positions are at the center, and 12 inches (centered) from the left and right inside walls of the hood. A scan of the edge or perimeter of the hood face was performed for the initial setup (denoted “Edge” in Table 1), with the detector probe hand-held and the mannequin removed. This setup was retested several times as indicated in Table 1. The graphs for the successful testing for some of the repeats show periodic small peaks, such as in Runs 101 Center, 101 Left, and 106 Left. It is believed that these are due to random fluctuations of room cross-drafts and currents. Similar graphs are shown in the book *Laboratory Fume Hoods: A User’s Manual,* G.T. Saunders, Wiley-Interscience, 1993. In particular, Figures 9.8 and 9.10 of the Saunders book, in which diffuser cross-drafts were being examined, resemble the present graphs.

When the exhaust flow setting was reduced to 50 Pa, the fume hood failed, meaning average concentration at one or more mannequin positions was > 0.1 ppm. Likewise with supply flows reduced to 1.8 Pa and exhaust back to 72 Pa, the fume hood failed.

Large failure was found for the initial configuration when both back and nearby front door of room were opened, presumably due to a strong cross-draft.

A wing was added to the lower supply vent to add flow inwards into hood at the bottom. The fume hood passed with the wing added and exhaust flow at 72 Pa and supply flows at 2.2 Pa.

Two special tests were performed with the mannequin arms in the hood downward at a 45 degree angle. These tests are not described in the ASHRAE 110 standard. Hands were approximately 1 ½ inches above lower supply vent. With the lower supply wing added, the fume hood passed. Without the wing (original setup), the hood failed. With the arms raised to horizontal, the hood passed.

One test was performed with a fan-generated cross wind of 100-200 fpm, which caused large hood leakage. This test is also not described in the ASHRAE 110 standard, and traditional fume hoods may also fail this test.

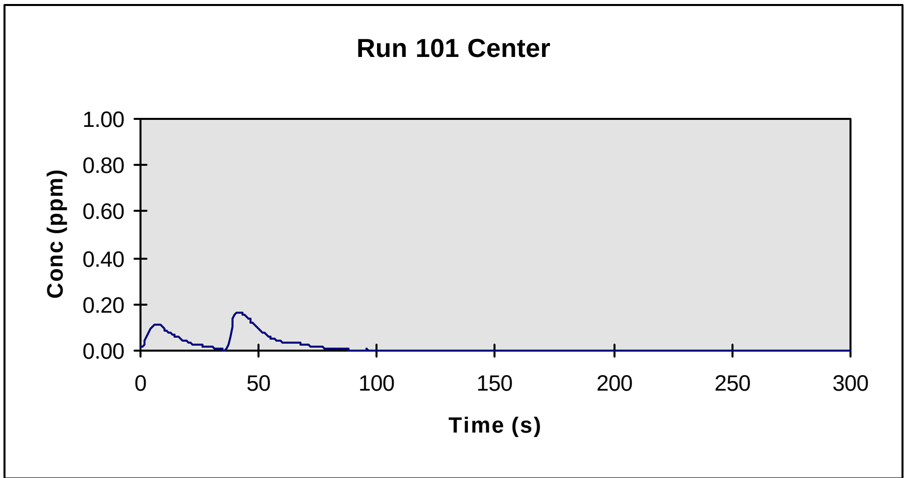
Graphs of tests with significant concentrations are shown following Table 1.

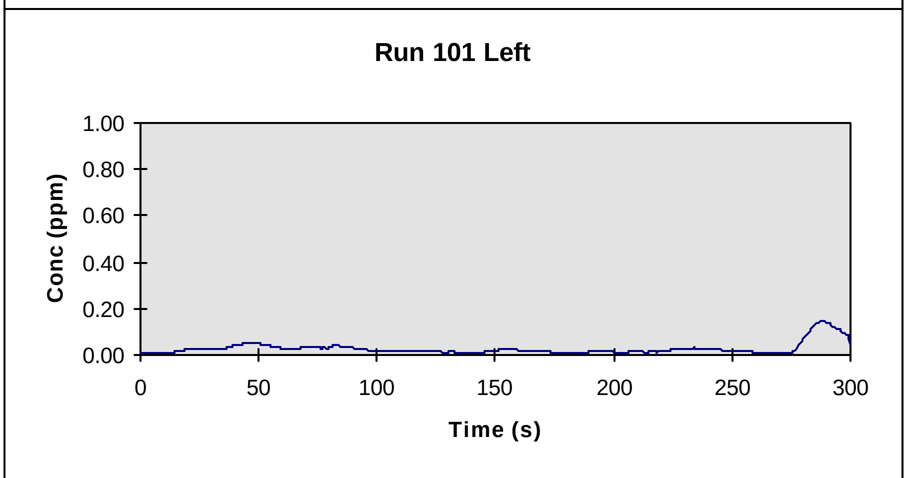
TABLE 1
Summary of Results
ASHRAE 110 Tracer Gas Tests

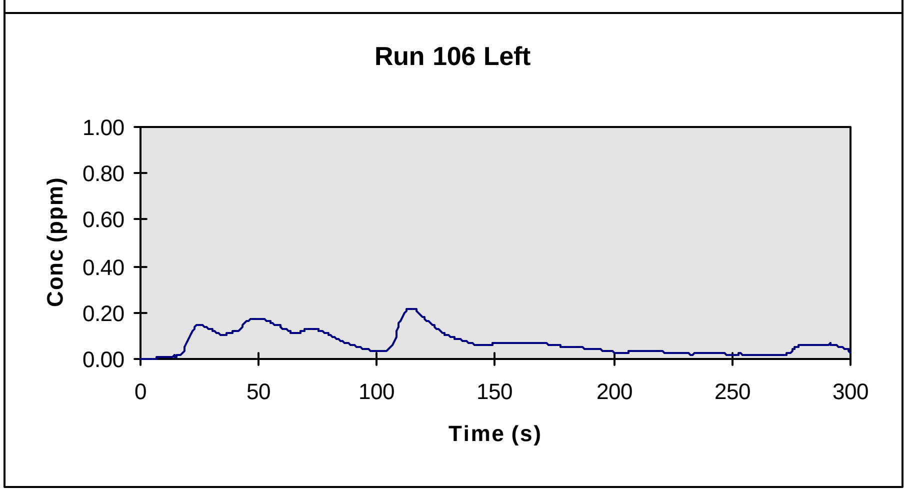
Lawrence Berkeley Laboratories
February 23-24, 1998
Ratcliff & Associates Project 00160

Run	Mannequin Position	PASS/ FAIL	Avg ppm	Max ppm	Comments
BASIC TESTS; exhaust=72Pa, Supply upper= 2.3Pa, lower=2.2Pa					
100	Center	PASS	0.001	0.013	
101	Center	PASS	0.015	**0.166**	door open
101	Right	PASS	0.000	0.003	
101	Left	PASS	0.027	**0.146**	
101	Edge	PASS	0.007	0.013	
106	Left	PASS	0.070	**0.219**	repeat
114	Right	PASS	0.009	0.027	door closed; 3 minute test
Effect of Exhaust Flow; exhaust = 50Pa					
102	Center	PASS	0.007	0.018	
102	Left	**FAIL**	**0.198**	**0.503**	
103	Left	**FAIL**	**0.140**	**0.305**	repeat of 102 Left
Effect of Reduced Supply Flow; (exhaust = 72Pa)					
104	Left	**FAIL**	**0.141**	**0.405**	upper=1.8Pa; lower=1.8Pa
105	Left	**FAIL**	**0.112**	**0.495**	upper=2.2Pa; lower=1.8Pa
Effect of Back/Front Door (exhaust=72Pa; supplies=2.2Pa)					
107	Left	**FAIL**	**0.171**	**0.995**	front door open/back open first half
108	Left	PASS	0.007	0.030	back closed/front open
Effect of Wing outlet lower supply (exhaust=72Pa; supplies=2.2Pa)					
109	Left	PASS	0.010	0.017	
109	Right	PASS	0.017	0.030	
109	Center	PASS	0.009	0.015	
Effect of Mannequin Arms at 45 degrees					
110	Right	PASS	0.023	0.067	Wing Outlet
113	Right	**FAIL**	**0.297**	**1.640**	No Wing Outlet
Effect of Mannequin Arms at 90 degrees					
111	Right	PASS	0.007	0.023	Wing Outlet
Effect of Strong Cross-draft					
112	Right	**FAIL**	**>3 ppm**	**>3ppm**	no data collection

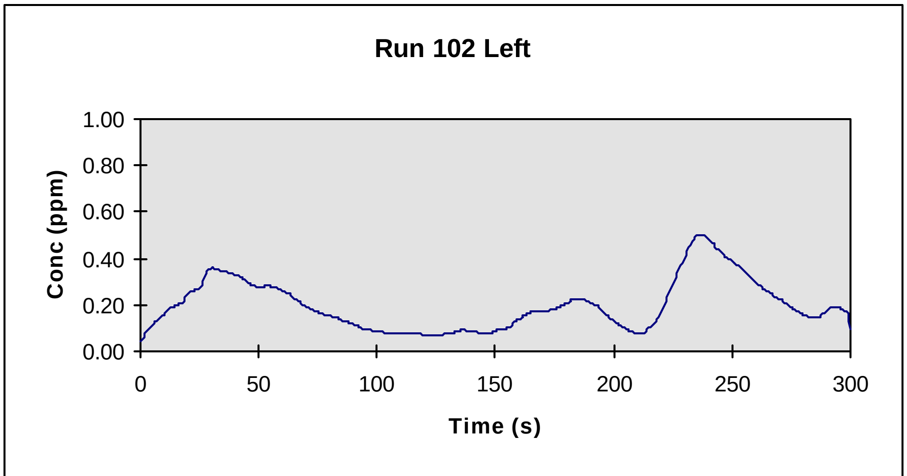
Graphs of Tests with Significant Concentrations; BASIC TESTS; exhaust setting= 72 Pa; Supply upper = 2.3 Pa; Suppy lower=2.2 Pa



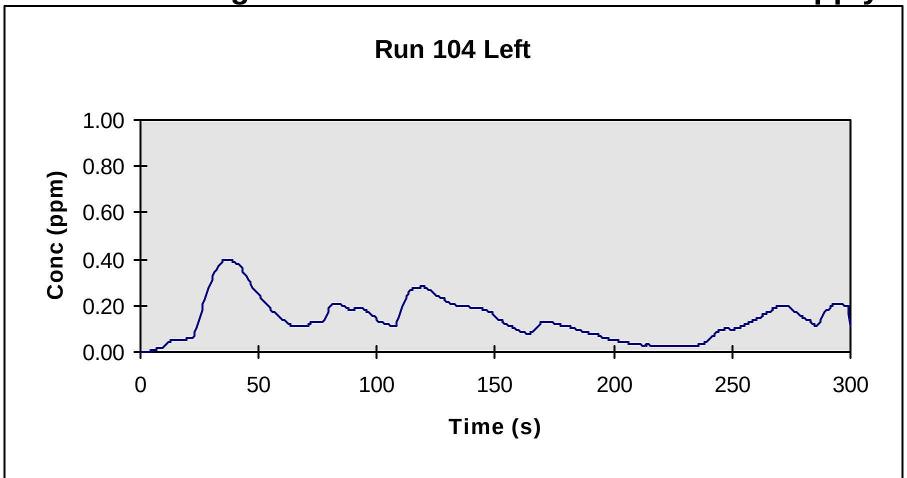


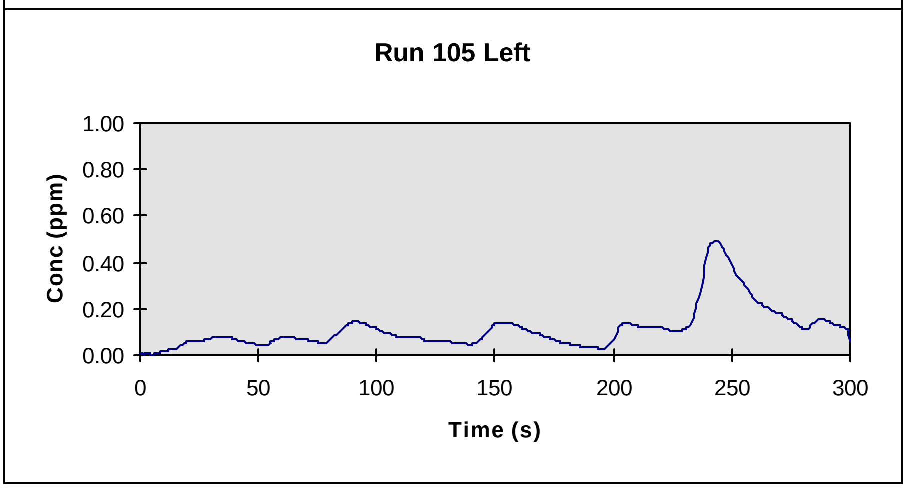


Graphs of Tests with Significant Concentrations: Exhaust = 50Pa.



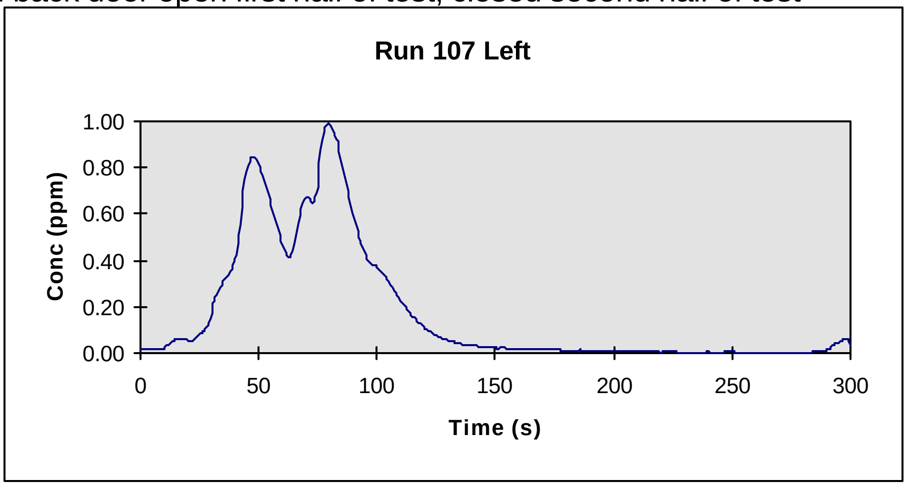
Graphs of Tests with Significant Concentrations: Reduced Supply Flow





Graphs of Tests with Significant Concentrations: Effect of Back/Front Door

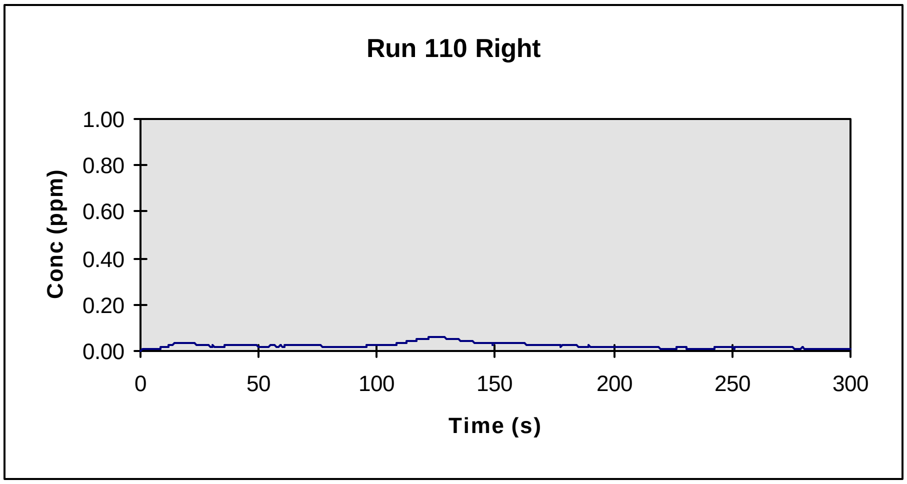
Run 107: back door open first half of test; closed second half of test



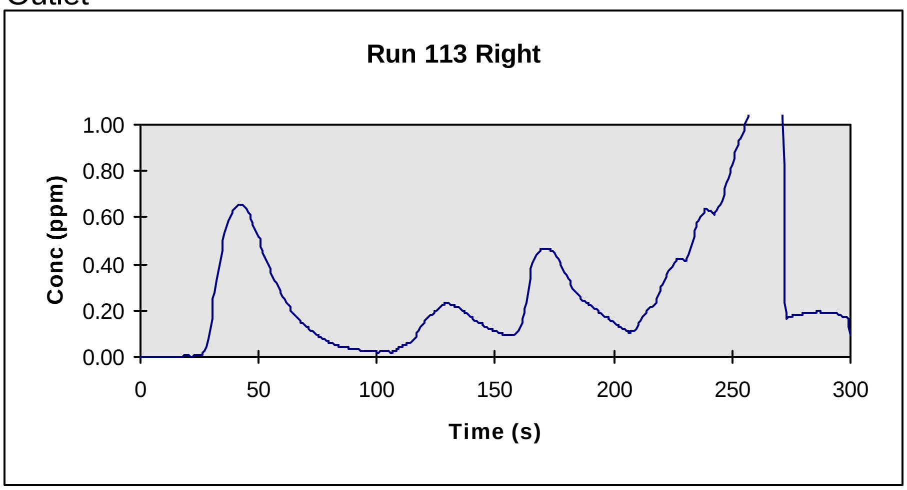
Back door closed; front open

Graphs of Tests with Significant Concentrations; Effect of Mannequin Arms at 45 degrees

Wing Outlet



No Wing Outlet



Thoughts to integrate:

a) supply at the top and exhaust at the bottom allow for bio hoods

b) maybe performance can be increased by exhausting also from the inner bottom grille and supply by the outer bottom grille.

c) low flows do not only save energy but also are easier to control as no VAV equipment has to be